

Performance of a beta-configuration heat engine having a regenerative displacer

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ABSTRACT

This paper investigates the performance of a beta-configuration heat engine having a regenerative displacer. In the conventional beta-engine; the displacer and the power piston are incorporated in one cylinder. The displacer transfers the working fluid between expansion and compression spaces via the heater, the regenerator, and the cooler. In the present work, successive homogeneous layers of square wire meshes occupy the displacer space of a beta-engine that make the displacer to be a displacer and a regenerator simultaneously. The theoretical analysis of the engine is based mainly on Schmidt theory. The optimum dimensions of the heater, cooler, regenerator, piston stroke and displacer stroke as dimensionless ratios of the bore were found. The optimum phase angle between the piston and the displacer and the optimum ranges of the speed for each working gas were also found. In a comparison between the proposed engine which has a regenerative displacer and the GPU-3 engine which has a stationary regenerator and a solid displacer; it was found that; the proposed one delivers 20% more power with 10% more efficiency than the GPU-3 engine.

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1. Introduction

The Stirling engine is an external combustion engine which uses compressible fluids as working media and is theoretically arranged to have Carnot efficiency. The thermal limits for its operation depend on the material used in its construction. In most instances, the engine operates at heater and cooler temperatures of 923 K and 338 K respectively, [1]. Fig. 1 shows the p - V diagram of the ideal Stirling cycle with complete regeneration. The rejected heat during processes from state 4 to state 1 is absorbed during processes from state 2 to state 3. An external heat sink and heat source are required during processes 1–2 and 3–4 respectively, [2]. The first Stirling machine was built and tested by Stirling (1816) as an alternative to steam engines and other heat engines. Since Stirling engines are externally heated, they are powered using wide varieties of fuels and heat sources. The Stirling refrigerator does not have ozone depletion potential, [3]. Fig. 2 indicates three different configurations; namely alpha-, beta- and gamma-configurations. Each one has the same thermodynamic cycle, but it has different mechanical design characteristics. The alpha-configuration has twin power pistons in two cylinders and the working gas fluctuates via the heater, the regenerator and the cooler. In the beta-configuration;

the displacer and the power piston are incorporated in the same cylinder. The displacer transfers the working fluid between the hot space and the cold space through the heater, the regenerator, and the cooler. The gamma-configuration uses two separate cylinders; one for the displacer and the other is for the power piston. The power piston cylinder is connected to the displacer cylinder by using a connection port, [4]. The regenerator effectiveness depends mainly on its surface area, its void volume and flow rate through it, [5–7]. The laminate regenerator is fabricated from stainless steel screen sheets that were stacked on top of each other at certain angular orientation. The wire mesh regenerator is assembled from successive layers of stainless steel wire meshes of different pores/inch. Heterogeneous arrangement of mesh layers enhances the regenerator effectiveness and reduces the pressure loss. A thermo hydraulic investigation on a packed bed solar heater having wire screen matrices of different geometrical parameters was mathematically modeled by [8]. The results indicated that the heater is thermo hydraulically efficient. The search for an engine cycle with high efficiency, multi-fuel and less pollution has led to reconsideration of Stirling cycle. Several engine prototypes were designed but their performances are still relatively weak if they were compared with those of combustion engines, [9]. This resulted from external and internal conduction, pressure drop in the regenerator and shuttle effect in the pistons. Current research indicates that solar powered, low temperature differential Stirling engines with gamma-configuration will be attractive in future, [10].

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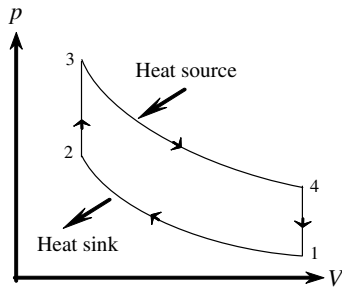


Fig. 1. Ideal Stirling cycle.

This paper investigates the performance of beta-type Stirling engine using square wire meshes as regenerative media in the displacer space. The proposed engine with the regenerative displacer is shown in Fig. 3, where the working fluid fluctuates through a reciprocating regenerator instead of the stationary one. The selection of beta-type engine is referred to its higher specific power than alpha and gamma types, [11,12]. The selection of wire meshes as regenerative media is due to the rigidity under high pressure, high durability, stability under expansion and compression and high regeneration effectiveness, [6]. The analysis of the engine depends mainly on Schmidt theory, [13], which deals with harmonic motion of the reciprocating elements, isothermal compression, isothermal expansion and perfect regeneration. The present study matched among the suitable dimensions of the heater, cooler and regenerator as dimensionless ratios of the cylinder bore. The piston stroke, the displacer stroke, the phase angle, the charging pressure and the speed of the engine were exhibited for six wire meshes.

2. Beta heat engine

The proposed engine consists of single cylinder with hot head at its top, power piston, cooler and regenerative displacer as shown in Fig. 3. The proposed engine has porous regenerative displacer instead of solid displacer and fixed regenerator that are existed in the conventional beta-engine. Thus, the displacer performs two jobs; the displacement of the working fluid between expansion and compression spaces and the regenerator job. This was identified as an idea for gamma engine by [11]. The reciprocation of the pistons is controlled by using the rhombic drive mechanism shown in Fig. 4. The piston and the displacer are connected to cross bars $c'b'$ and cb , respectively. The inter-meshing gears are used for the synchronization of the reciprocation, [14]. The position and velocity vectors of each link of the drive mechanism are expressed as follows:

$$\vec{r}_{ao} = (r_g - r)\cos\phi \vec{i} + r\sin\phi \vec{j} \quad (1)$$

$$\vec{v}_{ao} = \frac{d\vec{r}_{ao}}{dt} = -(r_g - r)w\sin\phi \vec{i} + rw\cos\phi \vec{j} \quad (2)$$

$$\begin{aligned} \vec{r}_{bo} &= \vec{r}_{ba} + \vec{r}_{ao} \\ &= [r_1\cos\phi_1 + (r_g - r)\cos\phi] \vec{i} + (r_1\sin\phi_1 + r\sin\phi) \vec{j} \end{aligned} \quad (3)$$

$$\begin{aligned} \vec{v}_{bo} &= \frac{d\vec{r}_{bo}}{dt} \\ &= [-r_1w_1\sin\phi_1 - (r_g - r)w\sin\phi] \vec{i} + (r_1w_1\cos\phi_1 + rw\cos\phi) \vec{j} \end{aligned} \quad (4)$$

$$\begin{aligned} \vec{r}_{b'o} &= \vec{r}_{b'a} + \vec{r}_{ao} \\ &= [r_2\cos\phi_2 + (r_g - r)\cos\phi] \vec{i} + (r_2\sin\phi_2 + r\sin\phi) \vec{j} \end{aligned} \quad (5)$$

$$\begin{aligned} \vec{v}_{b'o} &= \frac{d\vec{r}_{b'o}}{dt} \\ &= [-r_2w_2\sin\phi_2 - (r_g - r)w\sin\phi] \vec{i} + (r_2w_2\cos\phi_2 + rw\cos\phi) \vec{j} \end{aligned} \quad (6)$$

As x-component of $\vec{r}_{bo}$ is constant, i.e. $[r_1\cos\phi_1 + (r_g - r)\cos\phi] = r_3/2$; hence

$$\phi_1 = \cos^{-1}[\{r_3/2 - (r_g - r)\cos\phi\}/r_1] \quad (7)$$

And as a consequence, the y-component of $\vec{r}_{bo}$ which is the displacer movement y_b is:

$$y_b = r_1\sin\phi_1 + r\sin\phi \quad (8)$$

Similarly,

$$\phi_2 = \cos^{-1}[\{r_3/2 - (r_g - r)\cos\phi\}/r_2] \quad (9)$$

The y-component of $\vec{r}_{b'o}$ which is the piston movement $y_{b'}$, is:

$$y_{b'} = r_2\sin\phi_2 + r\sin\phi \quad (10)$$

The y-movements of both piston and displacer are represented in Fig. 5, where, one can get piston and displacer strokes and phase angle between them. Also, the swept volumes of both expansion and compression spaces can be stated. The variation of both expansion and compression spaces with the crank angle is expressed as follows:

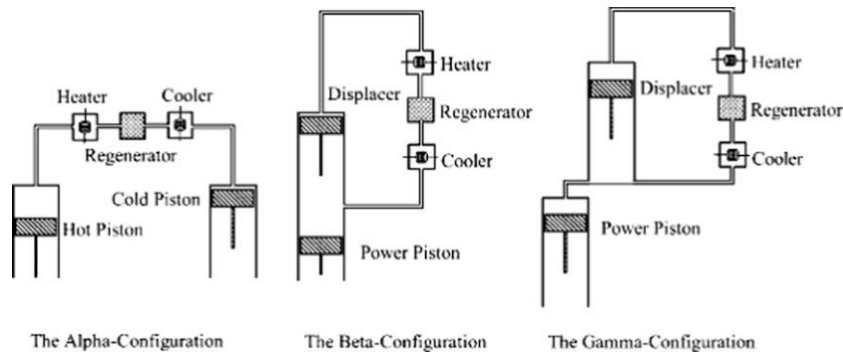


Fig. 2. Different configurations of Stirling engines. After [4].

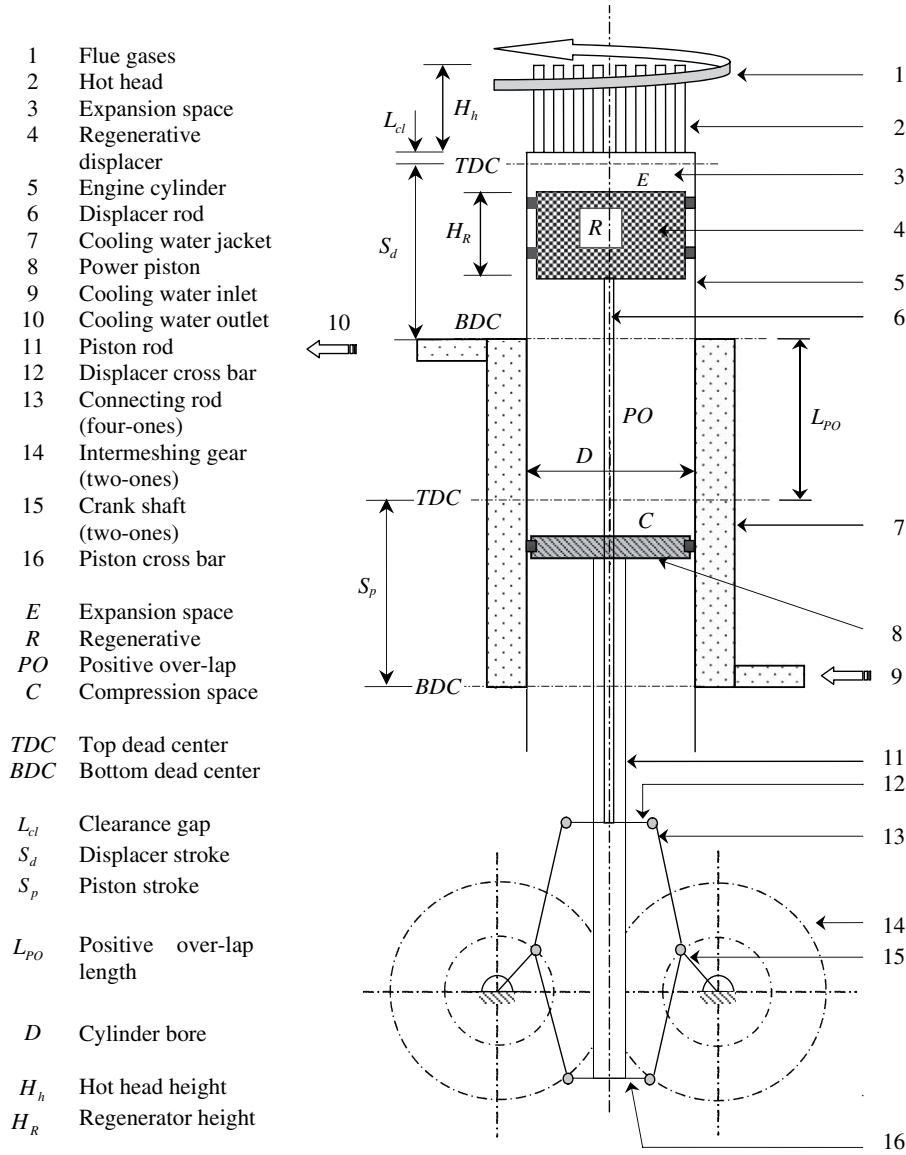


Fig. 3. Scheme of the proposed beta-type Stirling engine.

$$V_E = (V_{SE}/2) \times (1 + \cos \phi) + V_{cl,E} \quad (11)$$

$$V_C = (V_{SE}/2) \times (1 - \cos \phi) + (V_{SC}/2) \times [1 + \cos(\phi - \alpha)] + V_{cl,C} + V_{PO} \quad (12)$$

The engine workspace is divided into three isothermal regions in the vision of Schmidt theory, [13]. The expansion and heater spaces are at high temperature T_E , the compression and positive overlap spaces are at low temperature T_C and the regenerator space at an intermediate temperature T_R which is expressed as follows:

$$T_R = (T_E - T_C) / \ln(T_E/T_C) \quad (13)$$

According to Schmidt theory; the instantaneous pressure is kept constant throughout the engine workspace, and as a consequence, the mass of the gas is:

$$m_t = \frac{p(V_E + V_h)}{RT_E} + \frac{p(V_R)}{RT_R} + \frac{p(V_C + V_{PO})}{RT_C} \quad (14)$$

The Schmidt pressure is:

$$p = \frac{m_t RT_E}{\left[(V_E + V_h) + \frac{\ln(1/\xi)}{1 - \xi} (V_R) + \frac{1}{\xi} (V_C + V_{PO}) \right]} \quad (15)$$

The Schmidt p - V ellipse is represented in Fig. 6. Referring to Fig. 3, the mass conservation equation for each space of the engine workspaces is as follows, [15].

For the heater space;

$$dm_h/dt = 0 - \dot{m}_{h \rightarrow E} \quad (16)$$

For the expansion space;

$$dm_E/dt = \dot{m}_{h \rightarrow E} - \dot{m}_{E \rightarrow R} \quad (17)$$

For the regenerator space;

$$dm_R/dt = \dot{m}_{E \rightarrow R} - \dot{m}_{R \rightarrow PO} \quad (18)$$

For the positive overlap and compression spaces;

$$dm_C/dt = \dot{m}_{R \rightarrow PO} - 0 \quad (19)$$

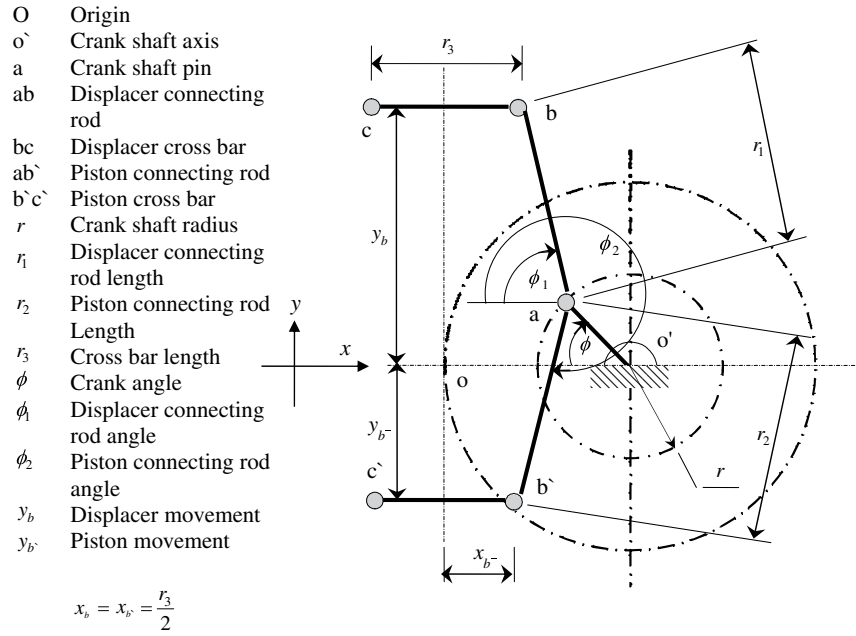


Fig. 4. Dimensions and angles of rhombic drive.

From equations (16)–(19); one can find; $\dot{m}_{h \rightarrow E}$, $\dot{m}_{E \rightarrow R}$ and $\dot{m}_{R \rightarrow PO}$. Hence the flow rate through the heater, the regenerator and the cooler can be found as follows:

$$\dot{m}_h = \dot{m}_{E \rightarrow h} \quad (20)$$

$$\dot{m}_R = (\dot{m}_{E \rightarrow R} + \dot{m}_{R \rightarrow PO})/2 \quad (21)$$

$$\dot{m}_k = \dot{m}_{R \rightarrow PO} \quad (22)$$

Fig. 7 shows the periodic flow rate through the heater, the cooler and the regenerator. The instantaneous Reynolds numbers through the heater, the regenerator and the cooler are found as follows:

$$Re_h = \dot{m}_h \times H_h / (\mu_E \times a_h) \quad (23)$$

$$Re_R = \dot{m}_R \times d_{hyd,R} / (\mu_R \times a_R) \quad (24)$$

$$Re_k = \dot{m}_k \times d_k / (\mu_C \times a_k) \quad (25)$$

2.1. Heater

The heater of the proposed engine is similar to the one which was experimentally tested by [16]. It is of inverted U-shaped cups that can be externally heated by flue gases as shown in Fig. 8. The experimental tests of such heaters concluded that the overall heat transfer coefficient during a complete cycle is; $U = 311 \text{ W m}^{-2} \text{ } ^\circ\text{C}$ for exhaust of LPG fuel at temperature $T_f = 600 \text{ } ^\circ\text{C}$. Thus;

$$Q_h = UA_{h,o} (T_f - T_E) \quad (26)$$

The pressure drop of the gas due to its flow through the heater is, [17]:

$$\Delta p_h = 1.6 \times \left[K \times \rho \times v^2 / 2 \right]_h \quad (27)$$

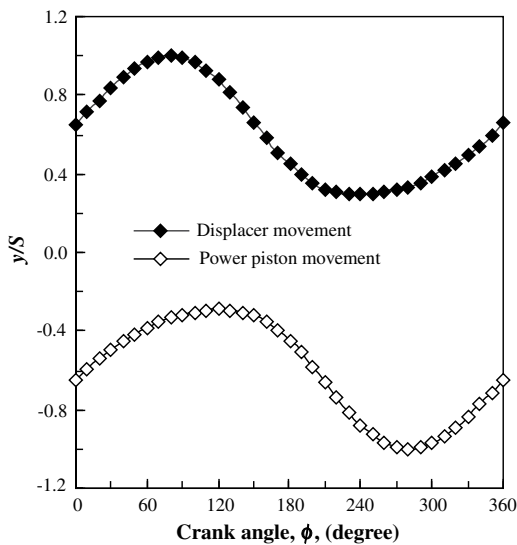


Fig. 5. Piston and displacer movement versus crank angle.

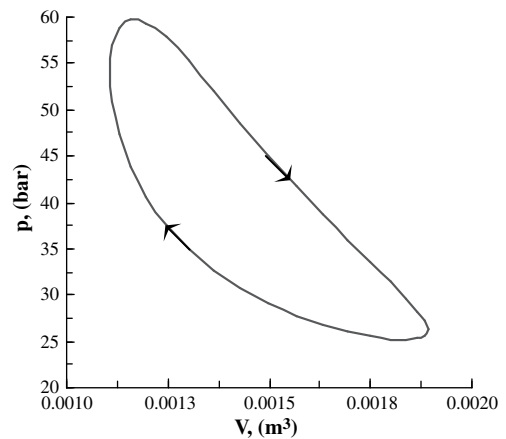


Fig. 6. Schmidt p–V diagram at a charging pressure of 20 bar.

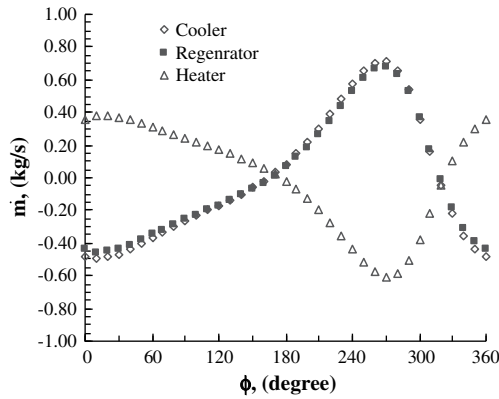


Fig. 7. Periodic mass flow rate through cooler, heater and regenerator.

2.2. Regenerator

The regenerative displacer has cylindrical shape, as shown in Fig. 8. It is formed of successive layers of stainless steel wire meshes having the same mesh size. The layers were homogenously stacked on the top of each other without revolving angle. Referring to Fig. 9, six meshes of different dimensions are suggested to be used in the proposed regenerative displacer; their technical data are given in Table 1, [6].

The pressure drop of the gas due to its flow through the displacer is, [18]:

$$\Delta p_R = \left[f \times (H/d_{hd}) \times \rho \times v^2/2 \right]_R \quad (28)$$

where

$$d_{hyd} = [\psi/(1-\psi)] \times d_w \quad (29)$$

The friction factor, f , is expressed as follows, [18]:

$$\log(f_R) = 1.73 - 0.93 \log(Re_R) \quad (30)$$

If $0 < Re_R \leq 60$

$$\log(f_R) = 0.714 - 0.365 \log(Re_R) \quad (31)$$

If $60 < Re_R \leq 1000$

$$\log(f_R) = 0.015 - 0.125 \log(Re_R) \quad (32)$$

If $Re_R > 1000$

The regenerator effectiveness is, [18]:

$$\varepsilon = NTU_R / (1 + NTU_R) \quad (33)$$

$$NTU_R = 2 \times \bar{St}_R \times H_R / d_{hyd,R} \quad (34)$$

and

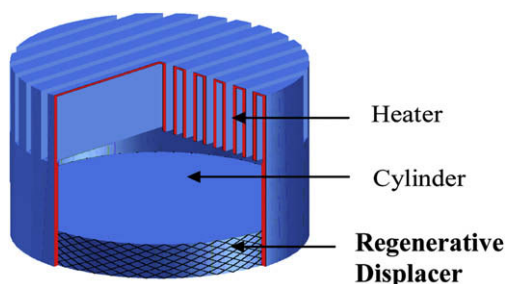


Fig. 8. Half cross-section in the heater and the cylinder head.

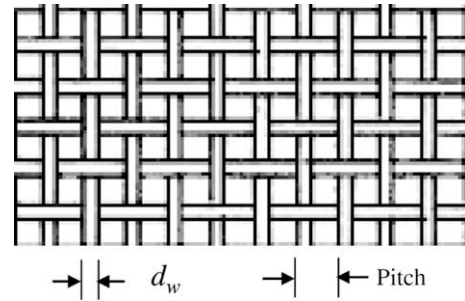


Fig. 9. Schematic diagram of wire mesh layer, After [6].

$$St_R = 0.595 / Re_R^{0.4} \times Pr_R \quad (35)$$

2.3. Cooler

The water jacket about the compression space is the cooler in the proposed engine. The working fluid is cooled during its fluctuation through the compression space. The cooling rate is, [19]:

$$\begin{aligned} Q_k^* &= h_o A_{k,o} (T_{k,o} - T_w) = 2\pi k_k L_k (T_{k,i} - T_{k,o}) / \ln(d_{k,o}/d_{k,i}) \\ &= h_i A_{k,i} (T_c - T_{k,i}) \end{aligned} \quad (36)$$

The pressure drop of the gas due to its flow through the cooler is, [17]:

$$\Delta p_k = 1.6 \left[(fL/d) \times \rho \times v^2/2 \right]_k \quad (37)$$

3. Engine power and efficiency

The elliptic p - V diagrams for expansion and compression spaces when taking the hydraulic losses into consideration are illustrated in Fig. 10. The indicated power is, [12]:

$$P_{ind} = N \times [\phi P_E dV_E + \phi P_C dV_C] \quad (38)$$

The rate of heat added is:

$$Q_h^* = N \times \phi P_E dV_E \quad (39)$$

The indicated efficiency is:

$$\eta_{ind} = P_{ind} / Q_h^* \quad (40)$$

A computer program in the form of a spreadsheet was prepared on the Microsoft EXCEL to compute the instantaneous volumes of expansion and compression spaces, instantaneous mass flow rates through the three heat exchangers, instantaneous Reynolds numbers in the heat exchangers, heat transfer rates, pressure drop, indicated power and efficiency. This was done for a complete revolution of the crank at a step of 1.0° . It was assumed that the

Table 1
Technical data of wire meshes.

Mesh- i	D_w (mm)	Porosity; ψ (%)
Mesh-50	0.200	69.08
Mesh-100	0.112	65.37
Mesh-200	0.050	69.08
Mesh-300	0.030	72.17
Mesh-400	0.030	62.89
Mesh-500	0.025	61.35

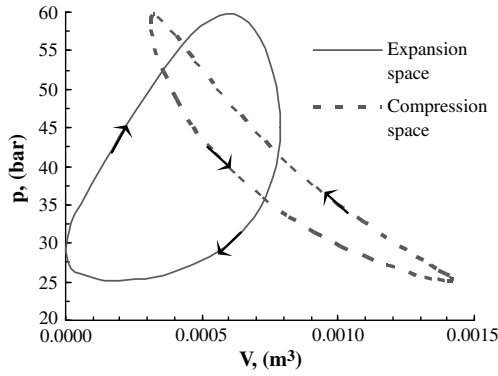


Fig. 10. Pressure-volume diagrams for both expansion and compression spaces.

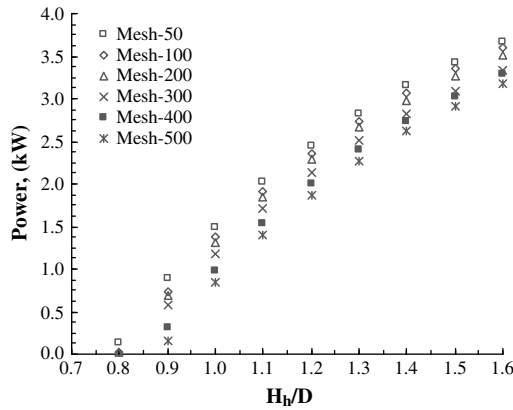


Fig. 11. Variation of power versus heater height for different wire meshes.

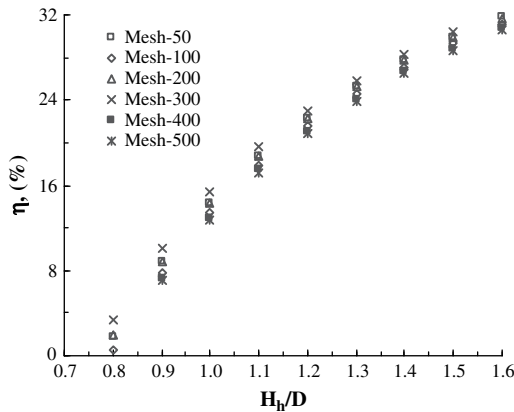


Fig. 12. Variation of efficiency versus heater height for different wire meshes.

engine works between temperature limits of $T_{\max} = 600^\circ\text{C}$ and $T_{\min} = 80^\circ\text{C}$. The cylinder bore is 100 mm. The program has the possibility to vary the dimensions of the heat exchangers, piston stroke, displacer stroke, phase angle, charging pressure and speed for each mesh size.

4. Results and discussion

The heater height to bore ratio, H_h/D was changed from 0.75 to 1.55 at a step of 0.1, the corresponding values of both power and

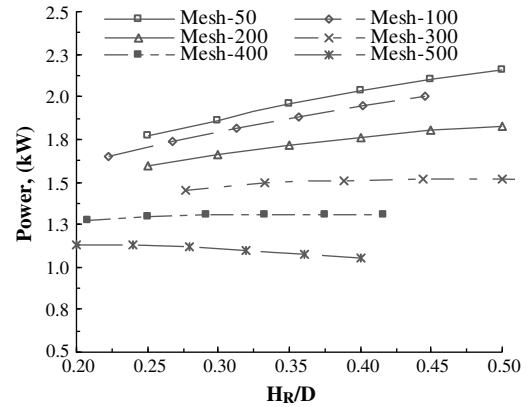


Fig. 13. Variation of power versus displacer height for different wire meshes.

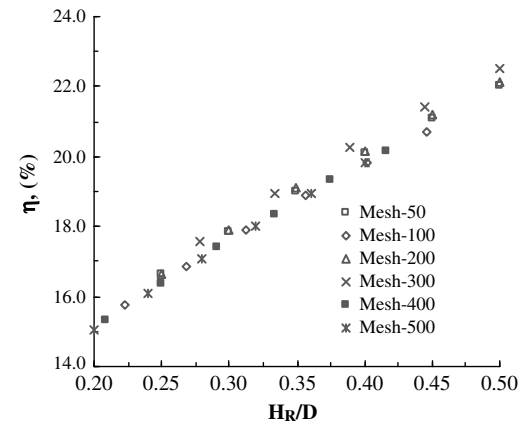


Fig. 14. Variation of efficiency versus displacer height for different wire meshes.

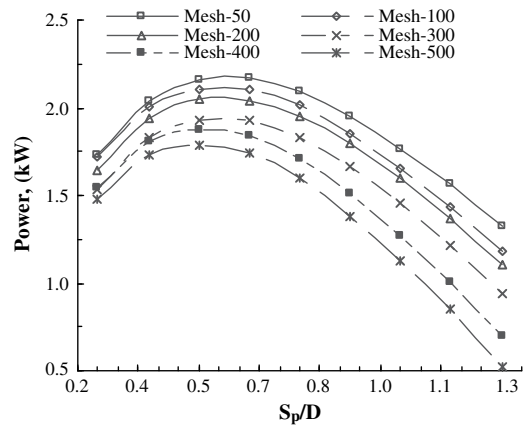


Fig. 15. Variation of power versus piston stroke for different wire meshes.

efficiency were determined. Fig. 11 shows higher rate of increase in the power with the increase of the heater height ratio up to about 1.0. Beyond this value, the rate of increase of the power reduces some how. This is referred to the increase in the heat transfer area; however, further increase in the heater height causes an increase in the dead volume which reduces the power. Also, thin wire meshes cause high pressure losses than the thick ones that reduce the power as shown in Fig. 11. The same tendency is even remarked in

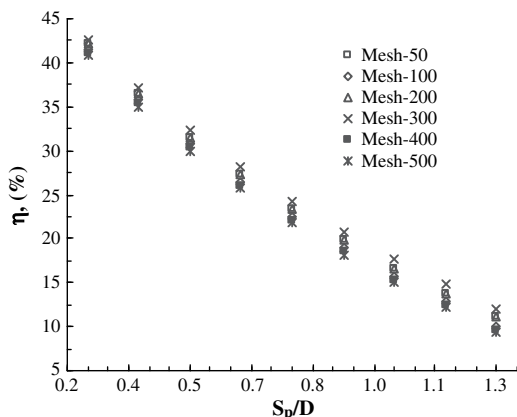


Fig. 16. Variation of efficiency versus piston stroke for different wire meshes.

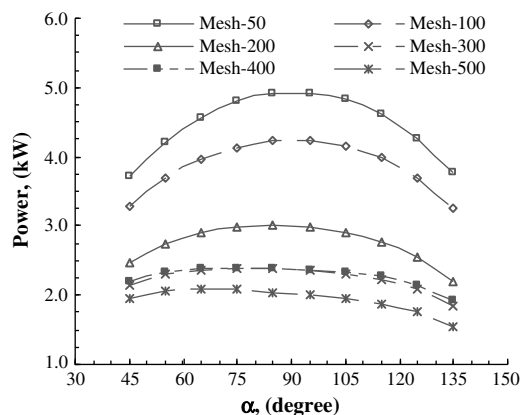


Fig. 19. Variation of power versus phase angle for different wire meshes.

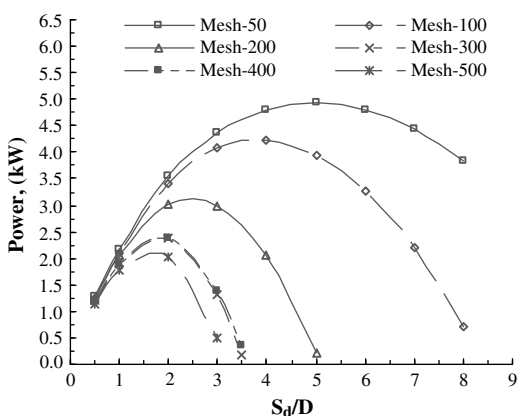


Fig. 17. Variation of power versus displacer stroke for different wire meshes.

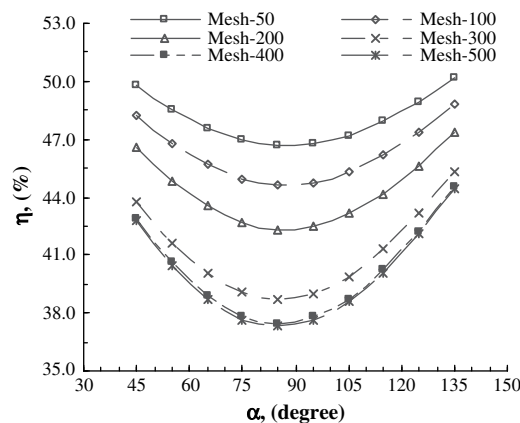


Fig. 20. Variation of efficiency versus phase angle for different wire meshes.

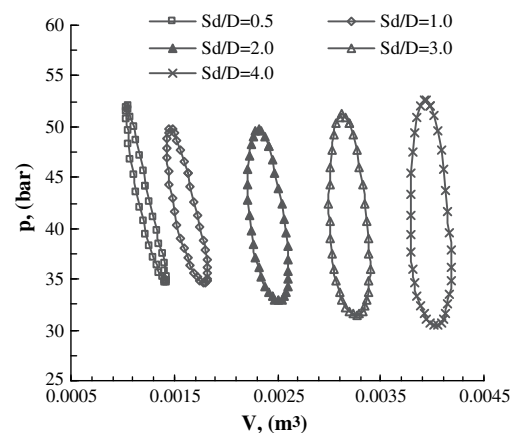
Fig. 18. Schmidt p - V diagrams at different displacer strokes for mesh 100.

Fig. 12 for the efficiency. Therefore; the heater height was selected to be $H_h/D = 1.0$ for every mesh size.

For the same displacer height, the thin wire meshes show more decrease in the power than that of thick ones as shown in Fig. 13.

Table 2
Optimum displacer stroke for each mesh size.

Mesh size	50	100	200	300	400	500
S_d/D	5	4	3	2	2	2

This is due to the increase in hydraulic losses. The increase in the displacer height increases the mass of the gas which increases the power. Also, the increase in the displacer height increases the regenerator effectiveness which increases the efficiency as shown in Fig. 14.

Increasing piston stroke increases the mass of the charged gas, which increases the indicated power up to about $S_p/D = 0.5$, as shown in Fig. 15. Beyond this value, the more increase in the piston stroke increases the charged gas which results in high pressure losses through the heat exchangers and as a consequence the power decreases. A reduction in the efficiency versus the piston stroke is noticed in Fig. 16, this is due to the increase in the heat rejection rate which resulted from the increase in the cooler surface area. Referring to Fig. 17, it is obvious that; each mesh size has an optimal value for the displacer stroke as given in Table 2.

Increasing the displacer stroke increases the charged gas which increases the power. Further increase in the displacer stroke increases the hydraulic losses, moreover; more increase in hydraulic losses was noted for thin meshes. So, the optimum displacer stroke decreases as the mesh size increases. The increase in the elliptic p - V area due to the increase in the displacer stroke is clear in Fig. 18.

The phase angle can be varied by the variation of the dimensions of the rhombic drive mechanism, [20]. The change in the phase angle causes a change of both compression and pressure ratios that affect the work per cycle and consequently the power. The phase angle was varied from 45° to 135° . Referring to Fig. 19, it is evident that; the more optimum phase angle for most of meshes is about 90° . The optimum phase angle results in high temperature ratio, ξ which reduces the efficiency as shown in Fig. 20.

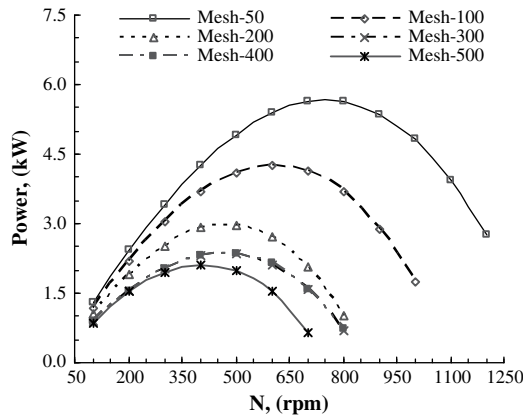


Fig. 21. Variation of power versus the speed for air as a working gas.

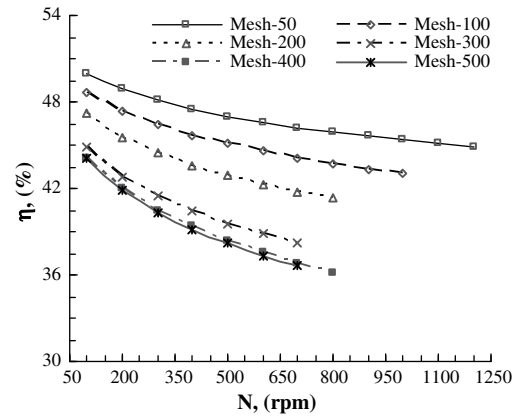


Fig. 24. Variation of efficiency versus the speed for nitrogen as a working gas.

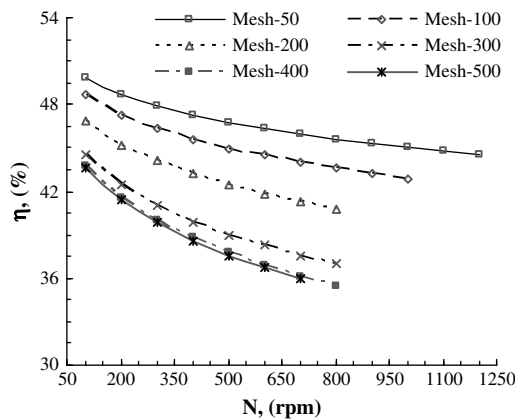


Fig. 22. Variation of efficiency versus the speed for air as a working gas.

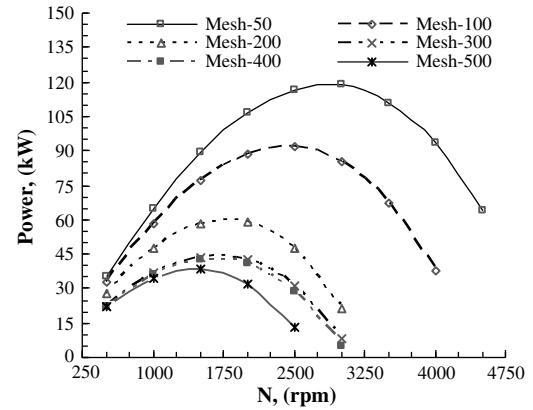


Fig. 25. Variation of power versus the speed for hydrogen as a working gas.

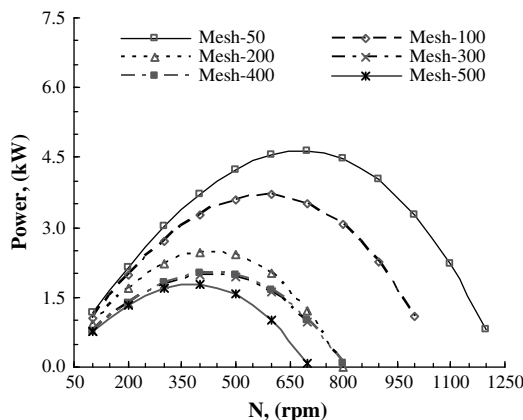


Fig. 23. Variation of power versus the speed for nitrogen as a working gas.

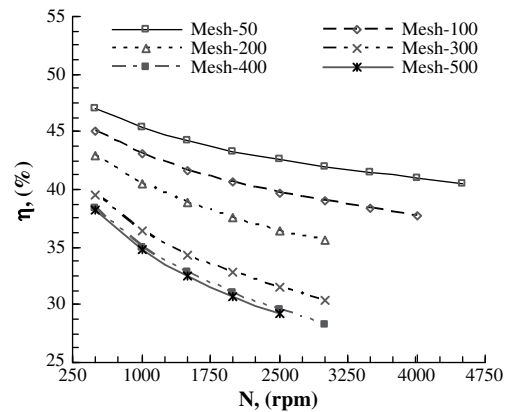


Fig. 26. Variation of efficiency versus the speed for hydrogen as a working gas.

Figs. 21,23,25 and 27 show the effect of the speed on the power if the engine was charged with air, nitrogen, hydrogen and helium respectively. The effect of the speed was obtained using the optimum dimensions of heat exchangers, optimum strokes and optimum phase angle for each mesh size. The optimum range of the speed for each gas has an acceptable range of the efficiency as shown in Figs. 22,24,26 and 28.

Regardless of the gas leakage and the lifetime of piston rings, one can say; the more charging pressure, the more power as shown

in Fig. 29. This is due to, the increase in the charged gas. The increase in the charging pressure causes a decrease in the regenerator effectiveness as shown in Fig. 29. This will increase the temperature ratio, ξ and as a consequence the efficiency decreases as shown in Fig. 30.

5. Comparison between present engine and GPU-3 engine

The urgent need to renewable energies has led to the development of Stirling engines that have an excellent theoretical efficiency, equivalent to Carnot. A previous modeling was applied on

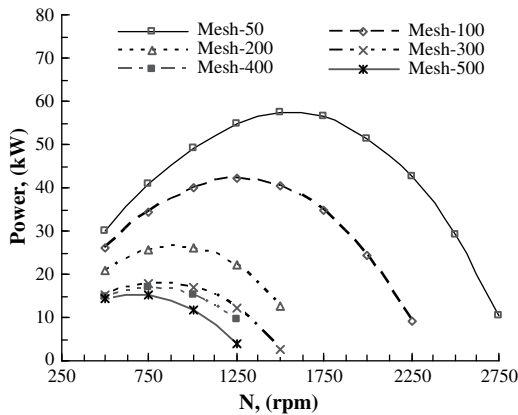


Fig. 27. Variation of power versus the speed for helium as a working gas.

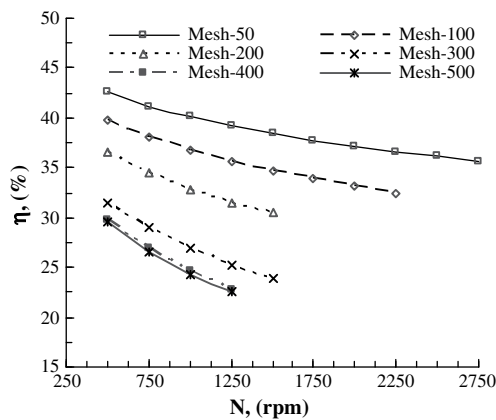


Fig. 28. Variation of efficiency versus the speed for helium as a working gas.

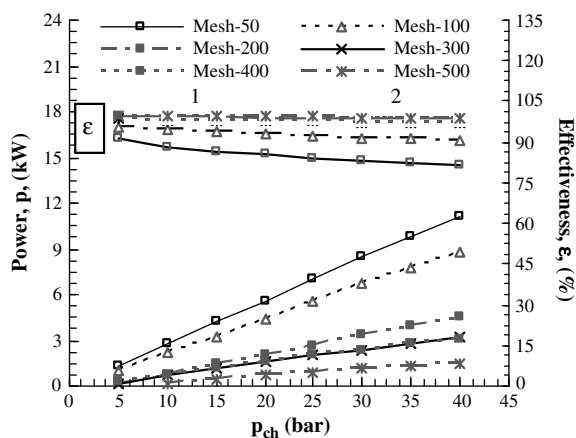


Fig. 29. Variation of power versus charging pressure for air as a working gas.

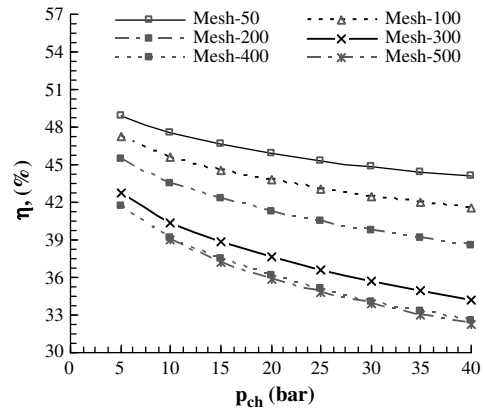


Fig. 30. Variation of efficiency versus charging pressure for air as a working gas.

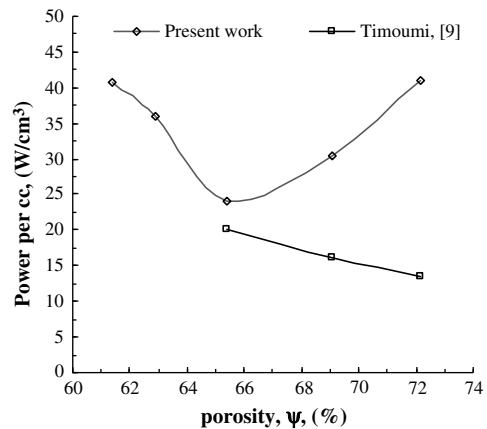


Fig. 31. Comparison between power per cc from present engine and GPU-3 engine.

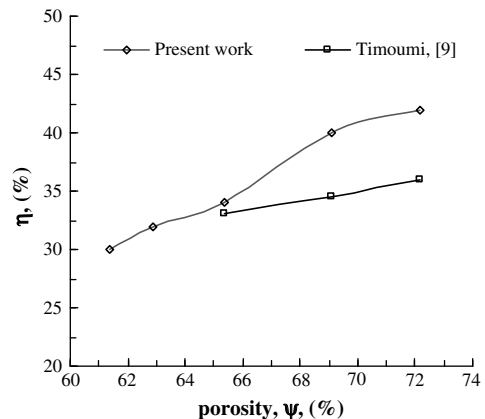


Fig. 32. Comparison between efficiency of present engine and GPU-3 engine.

6. Conclusions

The present work exhibited the performance of beta-type Stirling engine having regenerative displacer. Successive homogeneous stainless steel wire meshes filled the space of the displacer of engine. The porous displacer performs as a displacer and as a regenerator simultaneously. Six wire meshes of different mesh size were suggested to be the regenerative media. The analysis of the engine is mainly based on Schmidt theory. The optimum dimensions of heater, cooler, regenerator, piston stroke, displacer

the General Motor GPU-3 Stirling engine by [9]. The GPU-3 engine has a swept volume of 233 cm³ and uses helium at a mean pressure of 41.3 bar. The model considered $T_E = 977$ K and $T_C = 288$ K. The data of GPU-3 engine were fed to present model to calculate the corresponding results that were compared with those of [9]. The comparison shows that the present engine delivers about 20% more power per cc with about 10% more efficiency as shown in Figs. 31 and 32. As the porosity increases the present engine develops more power at high efficiency than that of the GPU-3 engine.

stroke as dimensionless ratios of the cylinder bore were found. The optimum phase angle and the optimum ranges of the speed for each gas were also stated. The main concluded remarks of the present work can be systemized as follows:

1. The proposed beta-engine which has a regenerative displacer instead of the solid displacer and the stationary regenerator delivers 20% more power per cc than that from the GPU-3 engine which has a stationary regenerator and solid displacer with an increase in the efficiency of about 10%.
2. The present model suggested successive layers of homogenous wire meshes of different mesh sizes. However, the model allows using of successive layers of heterogeneous meshes as well as other types of regenerative media.
3. The replacement of the stationary regenerator by a reciprocating regenerative displacer incorporated inside the engine cylinder of beta-type engines will reduce the space occupied by the engine and it will slightly reduce the engine weight.
4. The present model exhibited the dimensions of the engine as dimensionless ratios of the bore. And as a consequence, the engine sizes can be adapted to be similar prototypes.

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Appendix. Nomenclature

A	surface area, m^2
a	cross-section area, m^2
cc	cubic centimeter
D	cylinder bore, m
d	diameter, m
f	friction factor
H	height, m
h	heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$
i	wire mesh, pores/inch
$\vec{i}, \vec{j}$	Cartesian unit vectors
K	minor losses factor
k	thermal conductivity, $\text{W m}^{-1} \text{K}^{-1}$
L	length, m
m	mass, kg
$\dot{m}$	mass flow rate, kg s^{-1}
N	speed, rps
NTU	number of transfer units
P	power, W
p	pressure, N m^{-2}
$\dot{Q}$	heat transfer rate, W
R	specific gas constant, $\text{J kg}^{-1} \text{K}^{-1}$
Re	Reynolds number
$\vec{r}$	position vector, m
S	stroke, m
St	Stanton number
T	temperature, K
t	time, s
U	overall heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$
V	volume, m^3
v	velocity, m/s
$\vec{v}$	velocity vector, m/s
w	angular velocity, rad/s
x, y	cartesian coordinates, m
α	phase angle, rad
ε	regenerator effectiveness

ϕ	crank angle, rad
η	efficiency
μ	viscosity, kg m s^{-1}
ξ	temp. ratio = T_C/T_E
ρ	density, kg m^{-3}
ψ	porosity

Subscripts

C	compression space
cl	clearance
ch	charging
d	displacer
E	expansion space
f	flue gases
g	inter-meshing gear
h	heater
hyd	hydraulic
i	inner conditions
ind	indicated
k	cooler
o	outer conditions
p	piston
PO	positive overlap
R	regenerator
SE	swept expansion
SC	swept compression
t	total
w	wire

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